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PORTABLE MOMENT OF INERTIA MEASUREMENT DEVICE FOR BASEBALL BATS
FINAL DESIGN REVIEW

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ABSTRACT

This project was about designing a portable device that can measure the Moment of Inertia (MOI) of baseball bats to help players improve their swing. Right now, players only know a bat's weight and length, but MOI affects how a bat actually feels and performs. We built a lightweight, non-damaging system that accurately measures MOI and works with a wide range of bats.

Testing of the final device has not been completed yet, but the design shows a lot of promise based on initial analysis. Safety risks are minimal since the device uses a non-destructive method, but there could be minor risks if a bat isn't secured properly. With clear instructions and simple safety features, these risks can be controlled.

In the future, we would work on making the device even lighter and more automatic to improve ease of use and performance.

PROBLEM DEFINITION

The physical characteristics of baseballs bats available to players are typically just weight (in ounces) and length (in inches). The inertial properties of baseball bats can have significant impact on a player's swing mechanics, and are determined not only by weight and length, but can be more accurately defined by the mass moment of inertia (MOI). Currently, players have no way of knowing the MOI of the bats they feel comfortable using, so a method of reliably, accurately, and repeatably measuring the MOI of baseball bats is needed to optimize swing dynamics for a given player. Being able to determine the MOI of any bat can ensure even when using different bats, they all have consistent feel and not negatively impact the swing mechanics and satisfy player preference.

REQUIREMENTS, SPECIFICATIONS, DELIVERABLES

The device must satisfy the following functional and design requirements:

- Portability: The device must be lightweight and easy to transport.
- Bat Compatibility: It must accommodate a wide range of MLB bats, including variations in shape, length, and weight.
- Non-destructive Testing: The measurement process must not mark, scratch, or otherwise damage the bat in any way.
- Moment of Inertia (MOI) Measurement: The device must accurately measure the MOI of a bat relative to its major axis.
- Electronic Display: Measurement results must be displayed electronically in a readable format.
- Self-Contained Electronics: The device must operate using plug-in or battery power and must not rely on external computational devices.
- Measurement Repeatability: The system must yield consistent and repeatable results across multiple trials.

To ensure the requirements are met, the device must adhere to the following measurable specifications:

- Weight Limit: Device weight must be less than or equal to 20lbs. Validation of this specification will be weighing the device to confirm portability
- Max Bat Length: The device must be able to accurately measure a bat up to 40 inches in length. Validation of this specification will require testing with a 40 in cylinder and verifying the MOI measurement is consistent with shorter lengths.
- Max Bat Length: The device must be able to accurately measure a bat up to 40 ounces in weight. Validation of this specification will require testing

with a 40 oz cylinder and verifying the MOI measurement is consistent with lighter weights.

- Max Bat Diameter: The device must be able to accommodate a max barrel diameter of 3 inches. Validation of this specification will be done testing with 3 in diameter cylinder and ensuring measurements are able to be taken properly.
- MOI Accuracy: The device must be able to measure MOI within 5% error (one standard deviation) or 10% error (two standard deviations). Validation of this specification will take place by performing multiple MOI measurements on bats with known MOI and comparing to the data collected.

The final project will include the following components:

- Prototype Device: A functional, portable MOI measurement system meeting all requirements and specifications.
- Final Report: A comprehensive summary of design, methodology, testing procedures, and performance results, including statistical characterization of MOI accuracy.
- Theory of Operation Manual: A technical document explaining how the device functions, including underlying physics, electronic systems, and usage instructions.
- Detailed CAD Drawings with a Bill of Materials (BOM): Complete engineering drawings of all components, including a full list of parts used with quantities and sourcing information.

CONCEPTS

Please see the attached Figure 7 - Pugh Selection Matrix in the appendix. Below is a brief explanation of each possible design.

Design #1 (Figure 4) incorporated a pendulum concept that used a knife edge to balance the bat and bat holder. The knife edge serves to reduce points of contact and friction forces that would disrupt the pendulum motion. The motion is also recorded in this design by either a laser that records the passing of the bat every time it breaks the center plane, or photo gates that serve the same purpose. The downside of this design is that its bulky size would not allow it to be portable, and it also requires a controlled indoor environment. For the design to get accurate readings, the bat would need to be on a level surface, with no environmental interference like wind.

Design #2 (Figure 5) is a spring torque concept. The basic sketch in column two shows the bat holder being attached to a torsional spring which when loaded back into a stretched position, will apply a torque to rotate the bat. The bats rotational acceleration, recorded by an accelerometer placed over the axis of rotation, would provide the second variable, along with the known spring torque, to calculate the moment of inertia (I). The advantages of this design include its cost effectiveness, portability, and low risk of parts needing to be replaced, unlike in design three. For example: the spring alone costs less than $10.

Design #3 (Figure 6) uses the sane torque principle to get moment of inertia (I), but instead of using a spring mechanism it uses a motor. The disadvantages to using a motor are it is more expensive, it will need to be replaced more frequently, and it might get damaged or need regular maintenance. It will also be heavier than the spring mechanism, making it more difficult to transport.

Some changes did occur and updates from the initial Pugh Matrix selection were made. It was determined that the design for the bat holder would be moved from the original position of six inches from the handle to the center of gravity (CG) of the bat. This was done to eliminate gravity from the equation and the parallel axis theorem will be used to properly account for the difference in position. It was also determined that the easiest way to load the bat at the center of gravity is to incorporate a ‘V’ on which the user can find the balance point. Then the bat is secured in that horizontal position and the test is completed. This changed from originally testing the bat in a vertical position to testing it laying in a horizontal one.

MECHANICAL ANALYSIS

Error Analysis

As a dynamic device that relies upon user measurements, proper tolerancing to avoid friction, a notoriously finicky sensor (an accelerometer), and smooth, controlled motion, there is plenty of room for error propagation from device start-up to the final MOI measurement.

In such a case, even if the magnitudes of any single source of error can be mitigated, it is the accumulation of smaller imprecisions that can really devastate the reliability of a device such as this one.

For the current tolerance analysis, three main error sources have been considered: Center-of-mass position uncertainty, bat weight uncertainty, and accelerometer noise.

The balance point uncertainty comes from the assumed +/- 1 inch tolerance in finding the balance point when balancing the bat along the single “V” mount. It also comes from the subsequent imprecision in the user’s ability to measure the balance point value. The reading error was taken to be +/- 0.25 inches.

Next, bat weight is a quantity not measured directly by the moment of inertia measurement device; therefore, it is harder to estimate confidently. A safe assumption was taken of +/- 2 oz to account for variability in how one measures bat weight outside of using this device.

Finally, according to a source claiming to understand the measurement uncertainty of the LIS3DH triple-axis accelerometer when set to the range of +/- 2g, the accelerometer error was taken to be +/- 1.28 ft/s^2.

The presence of accelerometer noise was an aspect that offered a simpler opportunity for error reduction than the other more human-dependent error sources. Therefore, the MOI calculation program has been structured to take three instantaneous acceleration measurements at the time of release of the bat into motion. These values are then averaged to yield a more reliable result for the calculations. When N independent measurements are averaged, each with the same (random) uncertainty sigma, the uncertainty of the mean is reduced by a factor of sqrt(N), as shown in Eqn. (1).

$$\sigma_{\bar{x}} = \frac{\sigma}{\sqrt{n}} \tag{1}$$

Using this equation, the uncertainty due to accelerometer noise was reduced from +/- 1.28 ft/s^2 to +/- 0.739 ft/s^2.

Considering the above-mentioned sources of error, a complete error propagation analysis was achieved by assuming maximum uncertainty for average acceleration, balance point, and bat weight. In so doing, the moment of inertia under maximum error conditions was compared to the calculated moment of inertia under average conditions. The percentage error was found to be approximately 13.3%. The MATLAB code used to run the tolerance analysis can be found in the Appendix.

Fatigue Analysis

The spring used to move the rotation shaft in the internal assembly is a 40 in-lbs 302 stainless steel straight torsion spring. Using the specifications given on McMaster Carr, the supplier of the spring, an estimated fatigue analysis can be performed. The shear stress was first approximated by using Eqn. 2:

$$\tau = \frac{16M}{\pi d^3} \times K_t \tag{2}$$

With M (torque), d (wire diameter), and Kt (stress concentration factor) either given or estimated. The maximum shear stress was calculated to be 104 ksi. Using approximated values of 302 stainless steel and the torsional fatigue endurance limit (58 ksi), the fatigue factor of safety was calculated using Eqn. 3:

$$n_f = \frac{S_{e,torsion}}{\tau} \tag{3}$$

The calculated result is approximately 0.56, meaning that the spring does not have infinite life. However, the determining factor as to whether or not the spring should be used in the assembly is how many cycles of compression and decompression the spring can endure. Using the Basquin equation (Eqn. 4) for stress-life behavior or materials under cyclic loading:

$$\sigma_a = \sigma_f'\left(2N_f\right)^b \tag{4}$$

Solving for Nf/2, number of cycles until failure, an estimated fatigue life is calculated. For cycles ranging the full 90 degree range of motion, 6,600 cycles are possible until estimated failure. However, for the full assembly, the spring will only be cocked back to a maximum 80 degrees per cycle. At 80 degrees, the spring is estimated to survive up to 17,612 cycles. As shown in figure 1, the smaller the deflection angle of the spring, the more cycles until failure. Higher deflection results in higher stress and as a result, shorter lifespan for the spring.

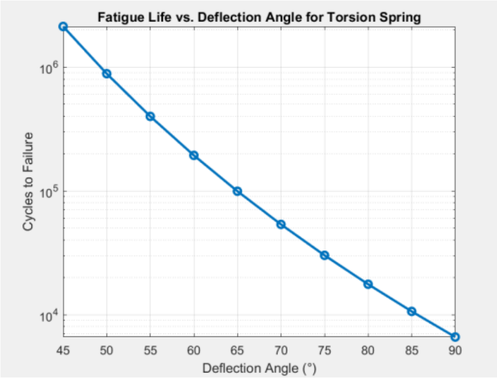

Figure 1: This graph shows the relationship between deflection angle and fatigue life for a torsional spring. As the deflection angle increases, the number of cycles to failure decreases exponentially. Operating the spring at lower deflection angles (e.g., under 60°) significantly improves fatigue life, with over 2,000,000 cycles possible at 45°, compared to 6,600 cycles at 90°. This analysis informed the design by ensuring the spring was used within a safe angular range to maximize durability and avoid early failure during repeated MOI measurements.

Fastener Torque Calculation

Three ¼" diameter alloy steel shoulder screws (10-24 thread, black oxide finish) are used in the device to constrain the arms of the torsional spring. The spring is mounted concentrically on a shaft, and its arms are restrained by the shoulder screws approximately three inches radially outward from the shaft center. Two screws are positioned above the spring arm and one below, forming a V-shaped channel that prevents the spring from rotating or slipping under load. The spring applies a torque of up to 40 in-lbf. To determine the force exerted on the shoulder screws, the relationship between torque (T), force (F = 40in-lbs) and distance (r = 3in) is used in Eqn. 5:

$$T = F * r \tag{5}$$
$$F = \frac{T}{r} = \frac{40}{3} \approx 13.3lbf$$

Thus, the max force the spring will exert on the shoulder screws which are about 3 inches from the shaft the spring rests on in about 13.3lbf. Since there are two screws on top and one on the bottom, arranged in a V-shaped pattern, the load is not perfectly symmetric but for simplicity and margin of safety, it was assumed each screw could see up to half the total force in the worst case scenario (which is much more likely for the top screws given the direction of the force from the spring arm). The calculations are done using Eqn. 6 below:

$$F_{per\ screw} = \frac{F}{2} \tag{6}$$
$$F_{per\ screw} = \frac{13.3}{2} \approx 6.7lbf$$

This calculation shows that each upper screw must be able to resist a tangential load of around 6.7lbf. The bottom screw although likely to see less/no load (more used for positioning) was also designed for 6.7lbf to ensure safety of the device. To ensure the screws were adequate, the shear strength was checked. With a shoulder screw diameter of 0.0250 inches and a minimum material shear strength of 84,000 [1] psi, the cross-sectional shear area (A) is determined using Eqn 7:

$$A = \Pi * \left(\frac{d}{2}\right)^2 \tag{7}$$
$$A = \Pi * \left(\frac{0.125}{2}\right)^2 = 0.0491in^2$$

And the maximum allowable shear force for each screw is:

$$F_{allow} = \tau * A \tag{8}$$
$$F_{allow} = 84000\frac{lbf}{in^2} * 0.0491in^2 \sim 4100lbf$$

Since the maximum applied load per screw is only 6.7lbf, each screw operates at less than 0.2% of its allowable capacity and ensures an extremely large safety margin against shear failure. To prevent loosening, an appropriate tightening torque was determined. The required clamp force per screw was assumed to be three times the applied force for safety purposes (Eqn. 9).

$$F_{clamp} = 3 * 6.7lbf \approx 20lbf. \tag{9}$$

Then using k = 0.2 (steel fasteners) [2] and d = 0.190in (10-24 major thread diameter) using Eqn. 10 below it is found the minimum tightening torque is ~0.75in-lbf.

$$T_{tighten = k * F_{clamp} * d} \tag{10}$$

Standard torque specifications for 10-24 shoulder screws recommend tightening to approximately 25in-lbf [3]. Therefore, a final tightening torque of 25in-lbf was selected to ensure secure, durable fastening under dynamic conditions. In addition to checking screw shear strength, thread pull-out strength in the aluminum mounting plate was analyzed (see Eq. #). With ¼-inch thread engagement (L) into the 6061 aluminum (shear strength ~18000psi [3]), each 10-24 screw (minor diameter of 0.159 inches) has an estimated pull-out strength of around 2250 lbf. With the shoulder screws bearing a maximum load of around 7lbf, the safety factor is well within safe range. See calculations below (Eqn. 11):

$$F_{pullout} = \Pi * d_{min} * L * \tau \tag{11}$$
$$F_{pullout} = \Pi * 0.159\ in * 0.25\ in * 18000\ \frac{lbf}{in^2} \approx 2250lbf$$

Initially, a smaller shoulder screw was going to be used in the design but after the calculations above and for safety purposes, it was decided to use the 10-24 shoulder screw. This analysis confirms that the shoulder screws used in the design are more than sufficient to safely constrain the spring arms under maximum expected loading without risk of failure or slip.

Discussion of Material Selection

When designing the bat hold section, the main goal was to keep the assembly light and make it easy to manufacture. The original idea was to use a light metal like aluminum, which is recognized as a light, strong metal. As the designing stage developed, the strength of aluminum for the bat hold proved excessive, as it is the heaviest a bat would ever be, about 3 lbs. With Delrin, tensile strength is about a third of the yield strength of aluminum (10000-11000 psi vs. 35000 psi) [8,9]. Accounting for the strength, a 1/2 inch of Delrin is enough not to deform any part of the Delrin during the swing. The density of Delrin is almost half of aluminum (0.0513 lb/in³ vs. 0.0975 lb/in³)[3,7], it allows Delrin to be a lighter solution for the bat hold but also provides for some pieces to be thicker in parts that may need it without being as heavy if it is necessary compared to aluminum. Delrin allowed lots of flexibility later in testing if any dimensional thickness required to be changed, while also allowing threads to be created like aluminum. Aluminum was chosen for the box plates and connecting bars for strength and stiffness purposes which did greatly impact the weight of the device compared to Delrin.

Discussion of Spring Sizing/Analysis

The spring for this part is employed to hold the bat onto the stationary V, and enough force is needed from the spinning V to keep it still while the swing is performed. The distance had to first be determined. To find this distance, the precise area where a bat will not be present and far enough away where the spin section will make a tight fit had to be located. The distance from the perpendicular position of the spin section's spring holder to the bat holder's spring holder had to be about 4-5 inches apart when looking at the NX CAD model. Assuming the spin section moves at most is about 45 degrees when looking at the CAD, this gives the minimum distance the spring would see, about 2.35-3.35in, which is a number that the spring must be more significant than to hold still enough tension to create a firm hold on the bat. To be safe, this maximum unstretched spring value will be 2-3in (Proportional to where the spring hold would be). With this in mind, a box of multiple springs with no values was given to us, where it was tested for these parameters. Since the range could be changed, looking for a spring within the set parameters and a high spring constant value (K) would be the best way to find the spring needed for this. To find K, use Hooke's law (Eqn. 12):

$$F = K(l - l_0) \tag{12}$$

Where l_0 is the initial length of the spring, l is the stretched length, and F is the force exerted. We used a push-pull spring scale to find the force (Eqn. 13) value by fixing one end and pulling on the spring scale side on top of a ruler until we got to a number that could be read, which was then related to the force exerted. For the spring picked out, it is 2.5in long unstretched, when pulled from 2.5in to 3in, the scale read 8.5lbf, which led to:

$$8.5 = K(3 - 2.5) \tag{13}$$

and led to finding K at 17 lbf/in. When setting the distance, it came down to changing the spring's smallest stretch size until there was a good value. 2.75in is the smallest stretch size for the spring when the spin section is at the closest point. To make sure the tension is strong enough to hold the bat, this distance between the spring poles was made to 4.5in when the spring section is set vertically, which is enough to hold the bat, especially when accounting for the friction from the rubber, which will stop the sliding.

Bearing Analysis

The smaller bearing connecting the bearing shaft and rotation handle was press fit into rotation handle. Proper tolerancing was required in order to ensure that the bearing was properly secured. Using a tolerance press fit chart, the 7/8" bearing is suggested to be pressed into a press fit hole of 0.8750 maximum diameter and 0.8745 minimum diameter. As for the larger bearing connecting the rotation shaft to the box wall, no tolerancing was used. Instead, during the turning process on the lathe, 0.005 in of material was removed at a time until the bearing was fit properly. A hole punch was also used to slightly increase the diameter of the aluminum shaft in order to allow the bearing to be pressed and secured into place.

In the design, a bearing was chosen to support the rotating shaft used in MOI measurement, but it is not being used in a traditional sense where it rotates continuously. Instead, the shaft rotates only slightly during testing and then returns to rest, meaning the bearing experiences very limited motion rather than sustained spinning. While bearings are typically designed for continuous/high speed rotation, they were selected over bushings in the application due to their low friction and more precise alignment. This helps to reduce measurement variability. Although a bushing could have functioned under these conditions, the bearing offers better repeatability and smoother motion even though it is operating outside its regular function.

Computer-Based Analysis

Finite Element Analysis (FEA) was performed to evaluate the effect of gravity-induced displacement on the baseball bat under different support conditions, with the goal of improving the accuracy and reliability of Moment of Inertia (MOI) measurements. The analysis simulated two scenarios: one in which the bat was held at the handle (top analysis), and another where it was supported at its center of gravity (bottom analysis). In both cases, a 10 N load and gravitational acceleration (9.81 m/s²) were applied to the bat geometry to simulate the physical forces present during testing. The results, shown in Figure #, clearly illustrate that supporting the bat at the handle resulted in significantly greater bending and displacement along the length of the bat. This deformation could negatively impact measurement accuracy and introduce risk of bending or damage during testing. By contrast, holding the bat at its center of gravity minimized structural deflection and effectively eliminated the contribution of gravity to rotational motion, allowing for cleaner, more accurate MOI readings.

Based on this analysis, the design was modified to support the bat at its center of gravity rather than at the handle. Holding the bat at the center of gravity effectively eliminated significant bending under its own weight, minimized displacement, and removed gravity-induced moments that could otherwise interfere with the accuracy of MOI calculations. Although additional simulations on varying bat geometries could further optimize support strategies, this initial analysis clearly demonstrated that supporting the bat at its center of gravity was the best approach for both accuracy and structural safety.

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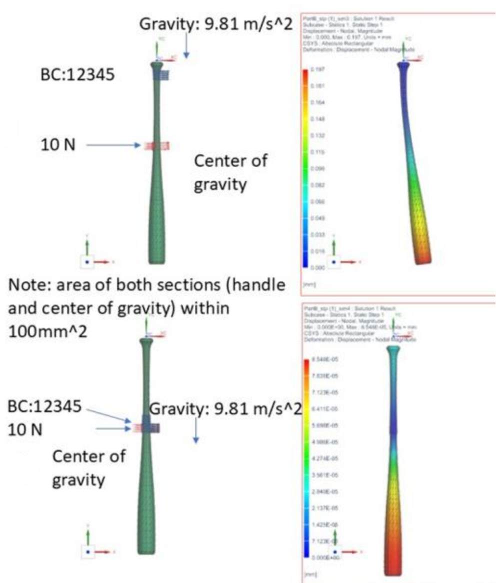

Figure 2: Finite Element Analysis of the effect of gravity and displacement if bat based on where it is held in device.'

Fundamental Mechanical Analysis

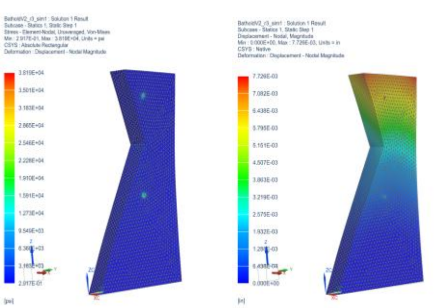

Figure 3: FEA of V-shaped bat hold to determine load distribution

An important mechanical analysis that was performed involved looking at how the geometry and material of the single "V" holding mount would stand up to the anticipated loads. To isolate this component in the analysis (shown in Figure 3) a few pieces of data were collected to understand the external loads applied to the mount. A force gauge was attached to the linear spring that constrains the double "V" mount around the sides of the bat to see how much force is applied. It was found that for a spring displacement of 2.5 inches, the spring force was approximately 17 lbf. However, NX simulation showed that a maximum case load condition was at about 45 degrees rotation of the double "V" mount. For this orientation, the displacement of the spring was found to be about 3.43 inches, which corresponds to a spring force of approximately 23.3 lbf. Therefore, the assumed force acting upon the single "V" due to the action of the moment pushing the bat into the single "V" was taken to be about 24 lbf. Furthermore, two points of contact were assumed between the cylindrical barrel of the bat and the triangular shape of the mount. For this reason, 12 lbf were assumed to act on the single "V" at either point of contact. The bottom surface of the "V", including the interior surfaces of the screw holes, were defined as fixed constraints. The analysis has further opportunity for improvement, such as modeling the bolt connections more accurately, and treating the load as a contact problem, not just a force applied at discrete points on the model. One way in which such analysis helped with design improvements was in realizing that the neck of the "V" could not be so thin. Please see the last page of the appendix for FEA analysis comparing Delrin to aluminum for the spinning "V"s.

MANUFACTURING

Manufacturing of the Moment of Inertia (MOI) measurement device involved a combination of 3D printing for early prototyping and traditional machining for the final components. Initially, a manufacturing readiness review (MRR) was conducted with Jim to evaluate the feasibility of the design and determine the best approach for machining critical parts. Sam Kriegsman provided significant assistance by reviewing CAD drawings and advising on manufacturability improvements, which helped streamline the fabrication process.

Early in the design phase, several parts were rapidly prototyped using 3D printing. This allowed for multiple CAD iterations and testing of part fits and function without committing to full machining. Once final designs were validated, parts were machined directly from raw materials, including Delrin and aluminum, for improved durability and strength. A variety of machines were used throughout fabrication, including a horizontal band saw for cutting raw stock, a manual lathe for cylindrical features, a table saw for large sheet cuts, and both a manual mill and Prototrak CNC mill for precision hole placement, slots, and profiles. Some components were fabricated on the spot during final assembly, with design adjustments made in real-time based on fitment testing. This agile approach was necessary due to delays in the arrival of raw materials, which compressed the manufacturing timeline into the final week before project completion.

Choosing to machine critical parts from raw materials instead of relying entirely on 3D printing was essential for meeting functional, dimensional, and strength requirements. While 3D printing was effective for initial fit and concept testing, machined parts provided the necessary precision, surface finish, and durability required for the final product.

TABLE 1
MANUFACTURING LABOR HOURS

Team Member	SCRUM Manufacturing Hours	Manufacturing Labor Cost (@ $100/hr)
PJ	15 hours	$1,500
Katie	22 hours	$2,200
Max	20 hours	$2,000
Total	57 hours	**$5,700**

The table below shows the number of hours put in by respective team members and the cost of the labor if we were paid at a rate of $100 per hour.

TABLE 2
TOTAL SCRUM HOURS

Team Member	Total SCRUM Hours
Katie	152 hours
Max	145 hours
PJ	138 hours
Zajans	165 hours
Team Total	**581 hours**

The table below shows the number of hours put in by respective team members throughout the entire semester in development and manufacturing.

Scaling for Mass Production

If the system were to be scaled to 1,000 units, several improvements could be made to significantly reduce cost and build time. First, parts currently machined from solid stock could be transitioned to lower-cost manufacturing methods such as injection molding for plastic components or die-casting for metallic parts. Standardized hardware and off-the-shelf components would be used wherever possible to minimize machining needs. Fixtures and jigs would be designed to allow faster, repeatable assembly operations, reducing manual labor time. Furthermore, tolerances could be slightly relaxed on non-critical parts to lower machining and quality control costs. Overall, by adopting mass production techniques and optimizing for manufacturability, it is estimated that unit production costs could be reduced significantly compared to the single prototype build.

TESTING AND RESULTS

Brief preliminary testing with both a baseball bat and aluminum cylinder have been completed. Results using a Louisville Slugger Genuine RA13CD Houston Astros bat has shown consistent result over 10 readings with a standard deviation of 20.09 oz*in2 with a percent error of 21.1%. Getting consistent data indicates reliable signal readings from the accelerometer and repeatability of the device measurements. Further adjustments to the system to reduce error from known values include:

- Verifying spring torque value
- Factoring mechanical tolerance analysis into calculations

Additionally, an aluminum tube (41 in, 43.3 oz) with a known MOI (6 in from end of tube) was tested to compare readings to known value and satisfy project specifications. See Table 3 below for comparison between the testing with the bat and aluminum tube.

TABLE 3
INITIAL TESTING DATA

Average MOI (oz*in^2)	Percent Error (%)	Standard Deviation (oz*in^2)
Louisville Slugger Genuine RA13CD Houston Astros bat		
8519.20	21.1	20.09
Hollow Aluminum Tube (41 in length, 43.3 oz weight) - Calculated MOI: 9207 oz*in^2		
10295.66	11.9	267.04

INTELLECTUAL PROPERTY

The portable Moment of Inertia (MOI) Measurement device for baseball bats is a novel system that addresses a critical gap in bat selection and player performance. To ensure that our design is original and does not infringe on existing intellectual property, a comprehensive patent search was conducted.

Several patents were reviewed that relate to MOI measurement, inertial properties of sports equipment, and similar testing devices:

- Patent US8944939B2 – "Inertial Measurement of Sports Motion" describes a system that measures inertial properties during active motion (e.g., a player swinging a bat). In contrast, our design measures MOI without requiring a player to swing the bat, but our design uses a torsional spring method, ensuring clear differentiation. [4]

- Patent US11833406B2 – "Swing Quality Measurement System" This system uses sensors mounted on the bat and requires player involvement. Our project avoids such complexity by focusing solely on the bat itself, enabling simple and repeatable measurements. [5]
- Patent JP2017023636A – "Swing Diagnostic Device and System" This Japanese patent relates to dynamic swing diagnostics. Again, our focus is on measurement without dynamic input from a player swinging the bat. [6]

Additionally, none of the patents involving bat swing analysis or inertial motion tracking (e.g., US9403077B2 or similar) match our design goals, which center on portability, low cost, minimal setup, and no player movement during testing.

Based on our prior research, our design does not infringe on existing patents and introduces key innovations:
1. A mechanical MOI testing approach without needing a player to swing.
2. Use of a torsional spring system instead of motorized or sensor driven designs.
3. Integration of simple mechanical elements (like a spring-loaded V-holder) to improve portability and robustness.

Several companies are currently active in related fields. Seiko Epson Corporation and Blast Motion Inc. focus on sensor based dynamic inertial measurements via a player's swing. Nike and Callaway Golf are a couple of the companies focused more on equipment fitting systems for swing dynamics.

Given the novelty and utility of this system, we believe the design and methos are patentable. Filing a utility patent could protect the specific mechanism of secure bat holding, static MOI measurement methodology, and the compact, portable device layout.

SOCIETAL AND ENVIRONMENTAL IMPLICATIONS

The development of a portable, accurate Moment of Inertia (MOI) measurement device for baseball bats has positive implications for public health and welfare, especially among athletes. By allowing players to select bats that are better suited to their swing dynamics, the device can help reduce strain and repetitive motion injuries commonly associated with poorly matched equipment. Improved swing mechanics can lower the risk of shoulder, elbow, and wrist injuries, especially in younger or developing athletes. Ensuring that players consistently use bats with similar inertia properties supports better long-term athletic performance and physical well-being.

There are minimal safety risks associated with the use of the device itself, given its non-destructive testing method and straightforward operation. However, if improperly secured during testing, the bats could potentially fall or shift, posing minor risks of injury. Proper usage instructions and design safety features (such as secure clamping mechanisms) can mitigate these concerns.

This device could have a big social impact too, especially since baseball is popular in many parts of the world. Right now, only players with access to a lot of equipment or expert coaching can really fine-tune their bat choice. Making MOI measurements easier and more available could help players from all backgrounds perform better. Its portability also means it could be used in lots of different places, from youth leagues to international tournaments.

The device does not have any ethical implications but coaches and players need to use the data honestly without trying to manipulate results.

RECOMMENDATIONS FOR FUTURE WORK

If we had an additional six months or another design cycle, we would focus on improving the system's overall accuracy and ease of use. Actions would be taken to try and reduce the size and weight of the device to enhance portability and make it more convenient for field use. Extended testing across a wider range of bats would allow the system to be fine tuned for maximum consistency.

To support these improvements, simulation tools such as Finite Element Analysis (FEA) could be incorporated into the design process. FEA could be used to model the spring wire constraint system, validate shoulder screw loading conditions, and predict contact stresses between the spring wire and the shoulder screws. Thread stress in the aluminum mounting plate could also be evaluated to ensure durability under cyclic loading. More broadly, simulation could help optimize the system's mechanical structure by identifying opportunities to reduce material use, lower weight, and improve stiffness without sacrificing strength. Integrating simulation-based design methods would allow a more systematic approach to enhancing the performance, reliability, and manufacturability of the device.

ACKNOWLEDGMENTS

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APPENDIX

Please see the following pages for extra figures, engineering drawings with a Bill of Materials and the tolerancing code. Also in the appendix are pictures of the design throughout manufacturing and the final design and FEA analysis of the spinning 'V's.

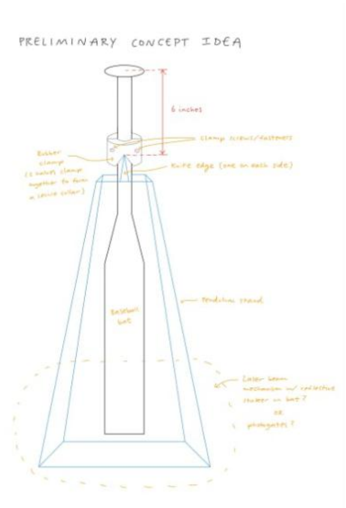


Figure 4: Pendulum Design (Design #1)

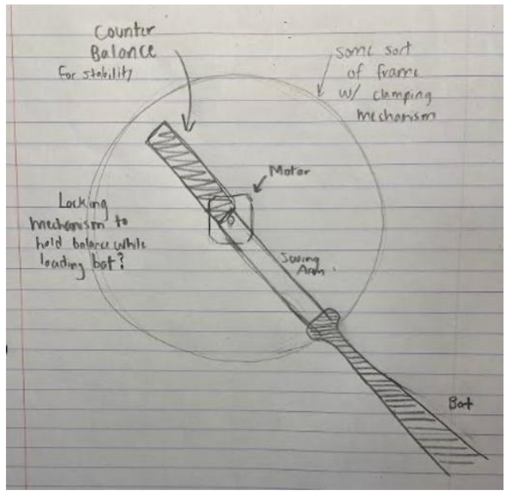


Figure 6: Motor Design (Design #3)

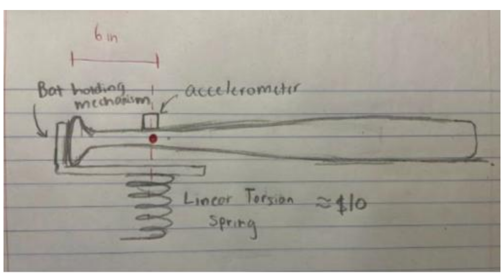


Figure 5: Spring Torsion Design (Design #2)

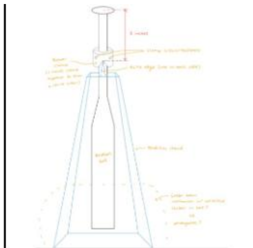
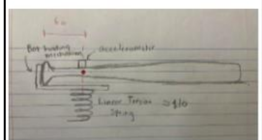
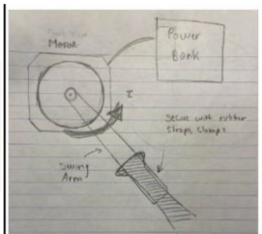
			
Cost	0	+1	-1
Weight	+1	+1	-1
Portability	0	+1	+1
Ease of Use	0	+1	+1
Total	+1	+4	0

Figure 7: Pugh Selection Matrix

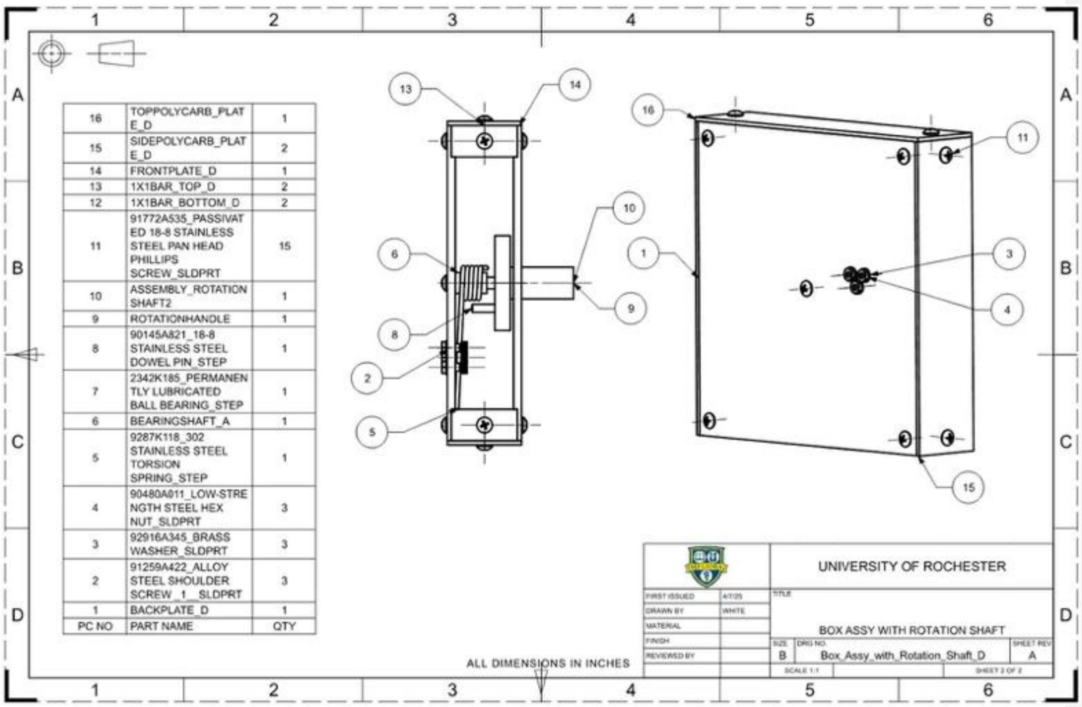


Figure 8: Box Assembly with Rotation Shaft

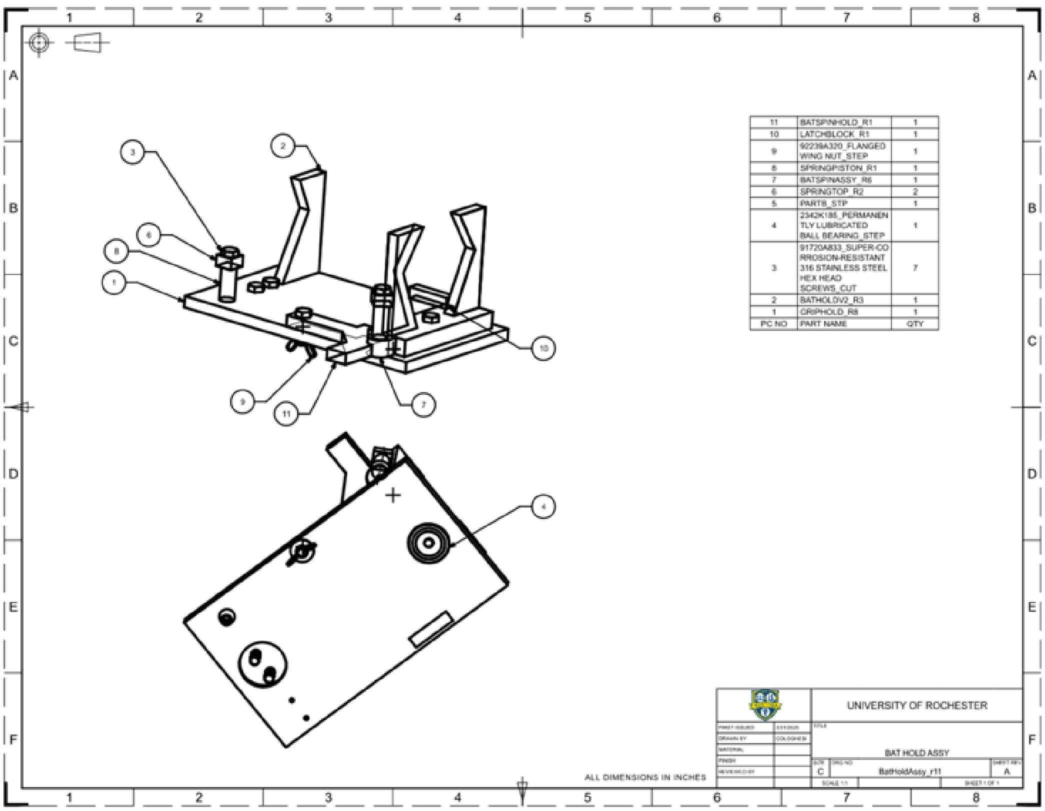


Figure 9: Bat Hold Assembly

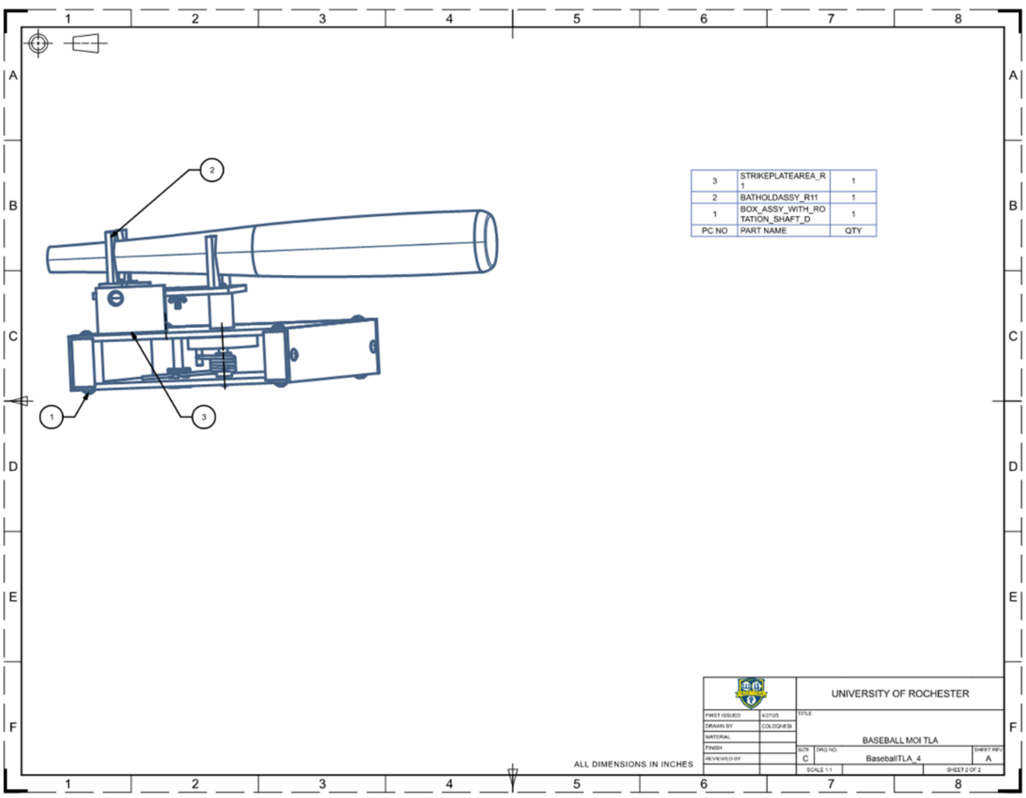
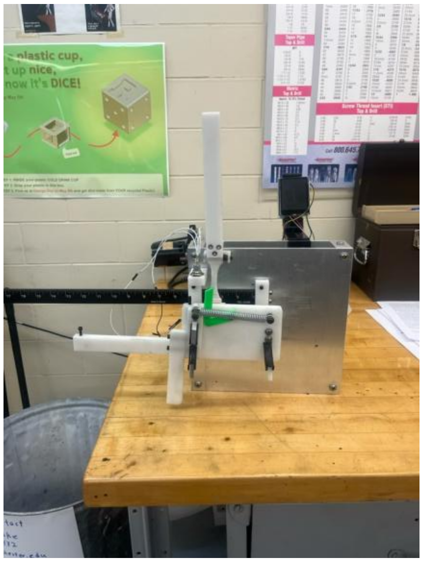
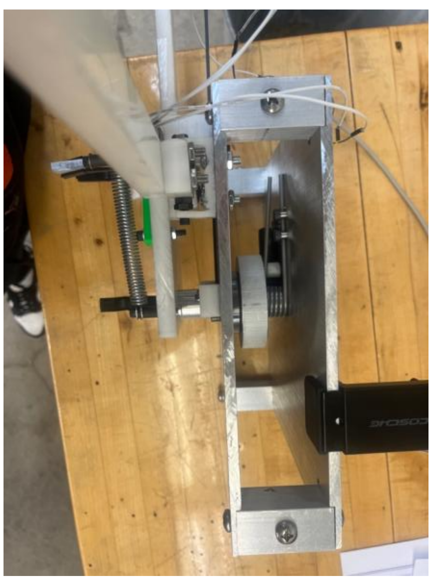


Figure #: Complete Assembly with Box, Rotation Shaft and Bat hold

Pictures of the device during manufacturing

Pictures of the final device





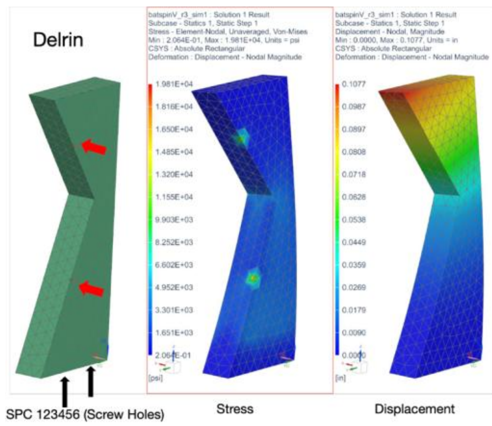
% Baseball bat MOI measurement device error/tolerance analysis

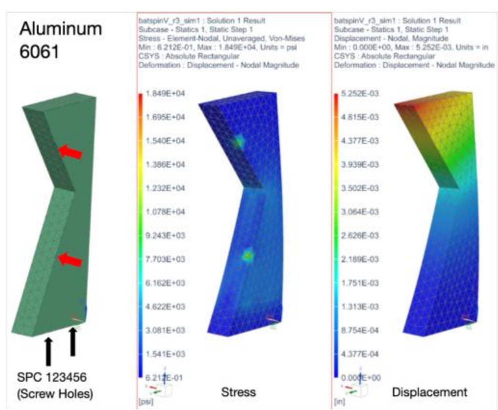
```
%% Baseball Error/Tolerance Analysis, Crapo, 4/28/25, Version 1.0
k_unitsPreferenceActivate('ft_lbf_s') % Defines unit system
k_unitsVariables('in, ft, lbf, deg, oz, s, slug, %') % Defines units used in calculations
% McMaster-Carr
% 302 Stainless Steel Torsion Spring
% 90 Degree Left-Hand Wound, 1.102" OD, 4.25 Coils
max_torque = 40*in*lbf; % Maximum torque
max_theta = 90*deg; % Maximum deflection angle
g = 32.2*ft/s^2; % Acceleration due to gravity
accelerometer_range = 2*g; % Accelerometer range
% T = torque
% k = torsional spring stiffness
% theta = deflection angle
% Assume linear torsional spring: T = k*theta
k = max_torque/max_theta; % Torsional spring constant
% alpha = angular acceleration
% I = MOI
% T = alpha*I
% alpha = (k*theta)/I
% Spring pre-load deflection angle: 70 degrees
% Spring pre-release deflection angle: 80 degrees
% (Bat gets cocked back by 10 degrees)
theta = 80*deg; % Actual deflection angle
% From "Swing Weights of Baseball and Softball Bats", Dan Russell, Kettering University, Flint, MI
% Link: https://www.acs.psu.edu/drussell/bats/Papers/TPT-October2010.pdf
% MLB-quality 34-in ash bat
% Pivoted 6 in from the end of the handle
pivot_point = 6*in; % Pivot point w.r.t. knob end of the handle
Io = 11239*oz*in^2; % MOI about 6 in pivot point
BP = 22.8*in; % Measured from the knob end of the handle
W = 31.2*oz; % Weight of bat in oz
m = W.convert('slug'); % Mass of bat in slugs
% Parallel axis theorem (assuming uniform composition of bat): Io = Icm + m*d^2
% Io = MOI of shape about point o
% Icm = MOI of shape about COM (same as centroid for uniform composition)
% m*d^2 = Added MOI due to distance between o and cm
d = BP - pivot_point; % Distance between 6 in pivot point and center of mass
Icm = Io - m*d^2; % MOI about center of mass
alpha = (k*theta)/Icm; % Angular acceleration
% a = linear acceleration
% b = distance of accelerometer from axis of rotation
b = 5.25*in;
a = alpha*b;
utilization_of_accelerometer_range = a/accelerometer_range;
percent_utilization_of_accelerometer_range =
utilization_of_accelerometer_range.convert('%');
% Bat weight error:
W_uncertainty = 2*oz; % The amount (+/-) that the measured weight can be off from the true weight
W_max_error = W - W_uncertainty;
m_max_error = W_max_error.convert('slug');
% Error propagation estimate: BP measurement + accelerometer measurement
```



```
BP_uncertainty = 1*in; % The amount (+/-) that the measured BP can be off from the true BP
% There is also error in how closely the user can read the measurement for the distance between the BP and the knob end of the bat
BP_read_uncertainty = 0.25*in; % How closely (+/-) that the user can read off the distance between the BP and the knob end of the bat
% Below is the source from which the uncertainty of the LIS3DH triple-axis accelerometer was found
% Link: https://web.cecs.pdx.edu/~gerry/class/IB2022/programming/on-board/accelerometer/
accelerometer_uncertainty = 1.28*(ft/s^2); % The uncertainty (+/-) of the LIS3DH accelerometer when the range is set to +/- 2g
% We are taking three instantaneous accelerometer readings and averaging these to get a more dependable acceleration value to use in the MOI calculation
% Therefore, when you average N independent measurements each with the same (random) uncertainty sigma, the uncertainty of the mean is reduced by a factor of sqrt(N)
N = 3; % Number of independent readings taken
average_acceleration_uncertainty = accelerometer_uncertainty/sqrt(N)
% Icm_max_error = T/alpha = T/[(a+average_acceleration_uncertainty)/b]
% Io_max_error = Icm_max_error+m*d^2 = Icm_max_error+m*[(BP-BP_uncertainty)-pivot_point)]^2
% MOI_percent_error = |(Io_max_error-Io)/Io|*100%
Icm_max_error = max_torque/((a+average_acceleration_uncertainty)/b);
Io_max_error = Icm_max_error+m_max_error*((BP-BP_uncertainty-BP_read_uncertainty)-pivot_point)^2;
MOI_percent_error = abs((Io_max_error-Io)/Io);
MOI_percent_error = MOI_percent_error.convert('%')
```

FEA Analysis of Spinning 'V's





Material selection for different parts of the device was done with stresses and high force areas in mind. Delrin, a lighter, easy to machine material was used for a large portion of the bat hold, however, aluminum was used for sections experiencing higher axial and bending forces. An analysis of the applied forces on the spinning bat hold section, seen above, uses 12 lbf loads applied at 2 points which are in contact with the bat after spring loading the device. When comparing FEM results of Delrin (left), the peak deflection of 0.1077in was much greater than the Al 6061 (right) with a peak deflection of 5.25E-3in. A deflection of 0.1077in is concerning, considering the "V" section is only 0.25in thick, making Al 6061 a much better material choice.